DESIGN AND DEVELOPMENT OF A GROUNDNUT PLANTER

By
BIMAL CHANDRA SAHU

A Thesis
submitted to the
Orissa University of Agriculture and Technology, Bhubaneswar
in partial fulfilment of the requirements
for the degree of

MASTER OF SCIENCE
(AGRICULTURAL ENGINEERING AND TECHNOLOGY)
IN
FARM MACHINERY AND POWER

DEPARTMENT OF FARM MACHINERY AND POWER
COLLEGE OF AGRICULTURAL ENGINEERING AND TECHNOLOGY
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C E R T I F I C A T E

This is to certify that the thesis entitled DESIGN AND DEVELOPMENT OF A GROUNDNUT PLANTER submitted in partial fulfilment of the degree of MASTER OF SCIENCE (Agricultural Engineering and Technology) in Farm Power and Machinery of Orissa University of Agriculture and Technology, Bhubaneswar is a faithful record of bonafide research work carried out by Sri Bimal Chandra Sahu under my guidance and supervision. No part of the thesis has been submitted for any other degree or diploma.

The help and information as have been availed of in course of this investigation have been duly acknowledged by him.

Sknanda
14.3.86
(S.K. Nanda)

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D) 14.3.86

Bimal Chandra Sahu

(BIMAL CHANDRA SAHU)

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INTRODUCTION

The present shortage of edible oil in the country has necessitated increasing the production of oil seeds in the country. Valuable foreign exchange is being spent to fill up the gap between indegenous production and demand. So during the Seventh Five Year Plan, Government of India have been taken up increasing the oil seed production the country. Therefore National Oilseed Development Programme is being instituted throughout the country. Under this programme farmers are being supplied with different inputs for production of oilseed crop at subsidised rates.

Groundnut is a major oil seed crop in India and also in Orissa. In India, production of Groundnut has increased almost two fold from 3.481 million tonnes in 1950-51 to 7.284 million tonnes in 1983-84 (6). The average per hectare yield of groundnut in Orissa is higher than the all India average (4). Annually 0.378 Million tonnes of Groundnut is being produced in Orissa in 0.278 million hectares in the State with an average yield of 13.58 quintals per hectare.

The increase in production has been possible through large scale adoption by farmers as a cash crop. Further increase in production of groundnut would be possible by adoption of high yielding varieties, balanced use of fertilizers, timely farming operations, reducing the labour and time and the cost of production. Mechanisation of the agricultural operations is the way to achieve this and make farming pleasant.

Groundnut can be grown in both Khariff and Rabi Seasons. In un-irrigated area it can be grown in the Khariff Season in the high lands. Also it can be grown in the flood plains and river bank alluvium in the coastal districts in the Rabi Season without irrigation. Where irrigation is available, groundnut can be grown after harvest of paddy, wheat or potato etc. Loamy, sandy loam and alluvial soil are the best suited for groundnut cultivation.

1.1. Scope and Justification

Groundnut being a root crop, the soil is well pulverized to facilitate plant growth. The normal practice of groundnut planting is by dibbling behind the plough. Groundnut is planted in rows 20 cm. apart with plant to plant spacing of 15 cm. For spreading varieties, the row to row spacing is 30 cm.

and plant to plant, 20 cm. The dibbling method is practised widely. Due to hand planting, the plant to plant spacing and depth of planting is not maintained. This affects the plant population and ultimately the yield.

Mechanical planting has not gained popularity. The planters available are costly and most suitable for big farmers. But in Orissa, land holdings are small. For small farmers, large bullock drawn or tractor drawn planters are becoming uneconomical to own. Therefore, keeping in view of the above situation, it was decided to develop a small manual drawn planter. This planter could be cheap, durable and can plant at least half a hectare a day.

In addition to being efficient this machine would relieve drudgery and make planting operation pleasant. This would not add to the unemployment problem already prevelant in the villages. Instead, it would attract young, educated, enthusiastic youth towards now un-attractive agricultural profession.

1.2 Objectives

The objective of the present research work is to -

i) Design and develop a low cost manually operated groundnut planter which would be simple in construction and operation and easy to be repaired by village artisans.

ii) Fabricate the different components of the groundnut planter and assemble them.

iii) To evaluate the performance of the groundnut planter.

Table : 1.1

Area, Production and Average Yield of Groundnut in India

Year	Area in Million Hectare	Production in Million Tonnes	Yield in Quintal/hectare
1950-51	4.494	3.481	7.8
1955-56	5.133	3.862	7.5
1960-61	6.463	4.812	7.5
1965-66	7.698	4.263	5.5
1970-71	7.326	6.111	8.34
1971-72	7.510	6.181	8.23
1972-73	6.990	4.092	5.85
1973-74	7.024	5.932	8.45
1974-75	7.063	5.111	7.24
1975-76	7.222	6.754	9.35
1976-77	7.043	5.264	7.47
1977-78	7.028	6.087	8.66
1978-79	7.433	6.208	8.35
1979-80	7.165	5.768	8.05
1980-81	6.801	5.005	7.36
1981-82	7.448	7.239	9.72
1982-83	7.215	5.282	7.32
1983-84	7.640	7.284	9.53

Source : Directorate of Economics Statistics, New Delhi.

Table : 1.2

Number of Operational Holdings of India as per Agriculture Census, 1980-81

Serial Number	Size group	No. of Holdings in million	Percentage (%)
1.	Marginal (below 1 ha.)	50.58	56.6
2.	Small (1 to 2 ha.)	16.10	18.0
3.	Semi-medium (2 to 4 ha.)	12.48	14.0
4.	Medium (4 to 10 ha.)	8.08	9.0
5.	Large (10 ha. & above)	2.15	2.4
Total		89.39	100%

Table : 1.3

Area and production of Groundnut in
Orissa, 1981-82.

Season	Area in Hectares	Production in tonnes	Yield in Quintal/ha
Khariff	1,05,976	1,01,891	9.6
Rabi	1,10,032	1,90,180	17.3
Total	2,16,008	2,92,071	13.5

Source : Agriculture Guide Book, 1985, Directorate of Agriculture & Food Production, Orissa, Bhubaneswar.

CHAPTER - III

DESIGN OF GROUNDNUT PLANTER

3.1 General

This chapter deals with the selection of the mechanism of the groundnut planter and the details of the design of each individual functional component of the planter.

3.2 Functional Requirements

The concept of a functional requirement of a manually operated groundnut planter is a pre-requisite for its proper design. These requirements of the machine are listed below.

(1) Power requirement of the planter should be low so that it can be pulled by one man in the field.

(2) The cost of the manufacturing should be kept as low as possible for individual ownership.

(3) The gross weight of the planter should be kept to a minimum because it has to be carried to the field by a single man.

(4) The design should be devoid of complicated mechanism as far as practicable.

(5) It should be able to be operated in the well pulverized fields and easy to handle.

of (6) The field losses due to planter should be minimum and the efficiency should be high.

i)

ii)

iii)

(7) Material of construction should be simple and readily available.

3.3 Power Requirement of the Planter

The planter is meant to be manually operated. Hence, the total power requirement should be limited to 0.1 h.p. or 74.6 watts. The draft required to pull the planter in the field should also be limited to 10 kgf. for effortless operation. The power requirement of the planter shall be divided between two major parts, namely -

i) pulling the planter in the field

ii) operating the metering device

3.4 Selection & Design of Metering Device

The selection and design of the metering device shall be done in three steps -

i) Selection of the type of metering device

ii) Dimension of the ground wheel

iii) Type of power transmission from ground wheel to metering plate.

3.4.1 Selection of the Type of Metering Device

Planters can have horizontal, vertical and inclined plate metering devices. In this case, the inclined plate metering has been selected due to the following reasons.

(1) This type has been used in planters since long and found to be satisfactory.

(2) Unlike horizontal plate metering, this does not require any seed cut-off device. Cut-off brushes may rub-off or damage groundnut seeds.

(3) In case of vertical plates, provision of small buckets are necessary in the metering plate. But in this case only notches can carry the seeds. Notches are easier to prepare and maintain than buckets.

3.4.2 Design of the ground wheel

In this planter, the ground wheel shall have two functions -

(1) To provide traction and power to the metering device.

(2) To act as press wheel to compact the soil above the seed.

The size of the ground wheel is governed by the size of metering plate.

The metering plate is taken to be 15 cm diameter having four notches. The spacing of planting for groundnut seeds is 15 cm.

So with 4 notches and taking 15 cm as the seed to seed spacing the distance the planter travels per revolution of the metering plate - 15 x 4 = 60 cm.

Diameter of Ground Wheel

Assuming a drive ratio of 1 : 1, the circumference of ground wheel = 60 cm.

Hence, the diameter of ground wheel = 60 ÷ 3.14 = 19 cm.

Ground clearance = 19 ÷ 2 = 9.5 cm.

This clearance is low.

Peripheral speed of the seed Plate

To calculate the peripheral speed of the metering plate, a maximum forward speed of 2 km. per hour is assumed.

$$\text{Speed of ground wheel} = \frac{2 \text{ kph}}{0.6 \text{ mm}}$$

$$= 3333 \text{ rev. per hr.}$$

Circumference of metering plate = 3.14 x 15 = 47 cm.

Speed of metering plate = 3333 r.p.h. x 0.47 m.

= 1.56 km/h.

This peripheral speed is quite high being higher than the upper limit for planters i.e. 1.38 km. per hour.

To reduce the peripheral speed of the metering plate, there are two alternatives -

i) the diameter of metering plate reduced

ii) the diameter of ground wheel increased.

If we reduce the diameter of the metering plate, the cups will come closer and there is change of multiple drops. So keeping the metering plate constant, the diameter of the ground wheel is increased which will reduce the peripheral speed of the metering plate as well as increase the ground clearance.

Selection of gear ratio

If the diameter of the ground wheel shall be increased, reduction of speed has to be done between ground wheel and the metering plate.

Employing one of the standard gear ratio of 1.6, the circumference of ground wheel = 60 x 1.6 = 96 cm

The diameter = 96 ÷ 3.14 = 31 cm

The peripheral speed of ground wheel at 2 km. per hour of forward speed = $\frac{2 \text{ km/h}}{96 \text{ cm}}$ = $\frac{2000 \text{ m/h}}{0.96 \text{ m.}}$

= 2083 rev/h.

= 34.7 r.p.m.

= 35 r.p.m.

Speed of metering plate

= (2083 x 0.47) m/h.

= 979 m/h.

= 0.971 m/h.

This peripheral speed is within the limits and hence the size of ground wheel and gear ratio is accepted.

The width of the ground wheel is taken to be 10 cms.

3.4.3 Power transmission from ground wheel to metering plate

The power from ground wheel to metering plate is to be taken in two steps.

i) First the power from the ground wheel is to be transmitted by roller chain and sprockets to an intermediate shaft parallel to the ground wheel axle.

ii) The power from this intermediate shaft is to be transmitted to metering plate shaft by a set of bevel pinion set having a gear ratio of 1.6.

3.5 Design of the Chain Drive

3.5.1 Selection of the Sprocket Pinion

The transmission ratio for the chain drive shall be 1 : 1. Hence the diameter and number of teeth of both the sprockets shall be same.

3.5.2 The Angle of Contact

It depends upon the centre distance between the sprockets and their diameters D_1 and D_2.

Minimum angle of contact is approximately 2.14 radians. The sprocket must be engaged by at least 5 or 6 chain links. To satisfy this requirement, the centre distance should have a value within the range discussed later in this chapter.

3.5.3 Maximum Allowable Velocity (V_{max})

Referring to standard tables by keeping the velocity range at about 100 rpm., and the power to be transmitted at 0.1 h.p. the chain 5.0S of IS : 2403 with the following specifications was selected.

Chain type	:	Roller Chain
Designation No.	:	IS 2403 5.0S
Rolon No.	:	G 52
Pitch (P)	:	8.0 mm.
Roller Dia	:	5.0 mm.
Width between inner plates	:	3.1 mm.
Pin body Dia (D_p)	:	2.3 mm.
Plate Depth (H)	:	7.04 mm.

Width over bearing pin	:	8.0 mm.
Bearing area	:	0.11 cm^2.
Breaking load	:	500 kgf.
Weight per metre	:	0.18 kgf.

3.5.4 Useful Force (P_1)

Maximum turning force P_{max} is obtained from the equation

$$P_{max} = \frac{\text{Ultimate strength of the chain}}{(\text{Service Factor } K_s)\ (\text{Safety Factor } K)} \qquad (3.1)$$

Here the ultimate strength of the chain was taken from the data available as 500 kgf. Safety factor K is taken to be 3, and service factor K_s is calculated from the equation.

$$K_s = K_1 . K_2 . K_3 . K_4 . K_5 . K_6 \qquad (3.2)$$

Where, K_1 = load factor = 1.25
(since load is variable with mild shocks)

K_2 = factor for distance regulation = 1.25
(since the centre distance is fixed)

K_3 = factor for centre distance of sprockets = 1
(since the centre distance of sprockets ap = 30 to 50 p.)

K_4 = factor for position of sprockets = 1
(since the inclination of the line joining the centres of the sprockets to the horizontal is within 60°)

K_5 = lubrication factor = 1.5
(since lubrication is periodic)

K_6 = Rating factor = 1.0
(since single shift operation of 8 hrs./day is proposed)

K_s = (1.25)(1.25)(1.0)(1.0)(1.5)(1.0) = 2.34

K = 3 aftg.

By substituting known values in equation (3.1)

$$P_{max} = \frac{\text{Ultimate strength of chain}}{K_s \cdot K}$$

$$= \frac{500}{2.34 \times 3} = 71 \text{ kgf.}$$

3.5.5 Peripheral velocity of the Chain (V)

The peripheral velocity of the chain is given by the following expression

$$v = p\, z_1\, N_1$$

where,
- p = pitch, mm.
- z_1 = number of teeth on sprocket pinion
- N_1 = speed of the sprocket pinion rpm.
- v = peripheral velocity, m/min

$$\therefore \quad v = \frac{(8.0)\,(22)\,(38)}{1000} = 6.688 \text{ m/min}$$

$$= 7 \text{ m/min.}$$

By knowing the peripheral velocity v, of the chain, the driving force can be calculated from the following equation

$$H.P. = \frac{F.V}{75} \qquad (3.3)$$

where
- F = driving force kgf
- V = peripheral velocity m/sec
- $H.P$ = power transmitted, P.S.

Assuming the full 0.1 h.p. of the manual power to be transmitted,

$$\text{Driving Force} \quad F = \frac{0.1 \times 75}{7/60} = 64.28 \text{ kgf}$$

The driving force (64 kgf) is much less than the ultimate strength (71 kgf) of the chain. So the chain is safe against failure.

3.5.6 The Pitch diameter (D_p) of the Sprocket

The pitch diameter of the sprocket is given by the expression

$$D_p = \frac{p}{\sin 180^\circ/z}$$

Where D_p = Pitch Diameter of the sprocket mm

z = Number of teeth on sprocket

p = Pitch mm

Here p = 8.0 mm

z = 22

$$\therefore D_p = \frac{8}{\sin 180^\circ/22} = 56 \text{ mm}$$

So the pitch diameter of the sprockets are 56 mm.

To ensure that with the number of teeth and pitch, the calculated diameter of the sprocket will be larger enough to be mounted on a shaft with due allowance for key way, the following equation is used for calculating minimum number of teeth on the sprocket.

$$z_{min} = \frac{4d}{p} + 5$$

Where d = bore diameter cm

p = pitch cm

z_{min} = minimum number of teeth

Substituting the values d = 13 mm., p = 56 mm

$$z_{min} = \frac{4 \times 13}{56} + 5 = 6$$

For fabrication a 22 teeth sprocket is taken readily.

3.5.6 Sprocket outside diameter (D_o)

The sprocket outside diameter D_o is given by the expression

$$D_o = D_p + 0.8\ d$$

Where D_p = Pitch diameter in mm

d = roller diameter in mm

Substituting the values of D_p = 56 mm

and d = 5 mm

In the above equation,

$$D_o = 56 + 0.8 \times 5 = 60 \text{ mm.}$$

3.5.7 Approximate centre to centre distance

The magnitude should always lie between a minimum (A_{min}) and maximum (A_{max})

$$A_{min} \angle A \angle A_{max} \qquad (3.4)$$

Where the value of A_{min} and A_{max} shown below

$$A_{min} = \frac{D_1 + D_2}{2} + 30 \text{ to } 50 \text{ mm} \qquad (3.5)$$

Where D_1 = Outer diameter of sprocket pinion

D_2 = Outer diameter of sprocket gear

A_{min} = Minimum centre to centre distance in mm

and A_{max} = 80 p (3.6)

Where p = pitch mm

A_{max} = Maximum centre to centre distance in mm

Substituting the value of $D_1 = D_2 = 60$ mm

$$A_{min} = \frac{60 + 60}{2} + 30 = 90 \text{ mm}$$

$$A_{max} = 80 \times 8 = 640 \text{ mm}$$
$$= 64 \text{ cm}$$

The actual centre to centre distance is 30 cm. so no problem arises regarding the minimum angle of contact.

3.5.8 Actual Factor of Safety (K_f)

This factor is obtained from the equation as follows

$$K_f = \frac{Q}{P_o}$$

Where Q = breaking load kgf

P_o = total force in the driving side of chain kgf.

Here Q = 500 kgf

$$P_o = P + P_c + P_s \tag{3.7}$$

Where P = net driving force in the driving side of the chain

P_c = chain tension caused by centrifugal force $= \frac{w}{g} v^2$

P_s = tensioning due to sagging $= k\,w\,a$

k = co-efficient of sag = 4

w = weight per metre of chain kgf

a = centre distance m

g = 9.81 m/sec^2

$$P = \frac{H.P. \times 75}{V_{peripheral}} = \frac{0.1 \times 75}{7/60} = 64.28 \text{ kgf}$$

$$P_c = \frac{w}{g} v^2 = \frac{0.18}{9.81} \times (\frac{7}{60})^2 = 0.00025 \text{ kgf}$$

$$P_s = 4 \times 0.18 \times x .30 = 0.216 \text{ kgf}$$

Substituting there values in equation (3.7)

$$P_o = 64.28 + 0.00025 + 0.216$$

$$= 64.496 \text{ kgf} = 64.5 \text{ kgf (say)}$$

Overall factor of safety $= K_f = \frac{Q}{P_o}$

$$= \frac{500 \text{ kgf}}{64.5} = 7.75$$

The forces acting on the wheel shaft

$$P_{shaft} = P_o + 2 k w a$$

$$= 64.5 + 2 \times 0.216$$

$$= 64.93 \text{ kgf}$$

3.5.9 Load on Shafts by the chain drive

The load exerted by chain drive on shafts is given by

$$Q_o = k_1 P_t \quad (3.8)$$

where Q_o = load on shaft due to chain drive kgf

K_1 = load factor = 1.15

P_t = tangential force due to power transmission kgf

$$P_t = \frac{(H.P.) \times 75}{v} = 64.5 \text{ kgf}$$

Substituting the values in equation (3.8) we get,

$$Q_o = 1.15 \times 64.5 = 74 \text{ kgf}$$

3.6 Design of Ground Wheel Shaft

The shaft of the ground wheel is connected to the drive mechanism by a chain sprocket having 22 teeth and outer diameter of 60 mm. It is fitted to the shaft by means of a screw. The sprocket speed = speed of shaft = 38 rpm.

The total force acting on the driving side of the chain can be resolved into two components.

(i) Horizontal component acting at 90° with the alignment of the shaft

and (ii) the vertical component.

From Fig. 3.1

$$F_h = P_o \cos 30^\circ = 64.5 \times 0.866 = 55.86 \text{ kgf}$$

$$F_v = P_o \sin 30^\circ = 64.5 \times 0.5 = 32.25 \text{ kg}$$

Taking moment about the axis of the shaft

$$M_h = 55.86 \times \frac{60}{2 \times 10} = 167.58 \text{ kg cm}$$

$$M_v = 32.25 \times \frac{60}{2 \times 10} = 96.75 \text{ kg cm}$$

Bending moment to due to the self weight of the shaft and the ground wheel is to be calculated considering the shaft to be fixed at both the ends and subjected to torsional force.

$$\text{So the } B.M_{max} = \frac{W.L}{24} \text{ at } x = \frac{L}{2}$$

$$= \frac{7 \times 10}{24}$$

$$= 2.92 \text{ kg cm} = M_{v\,2}$$

The net $M_v = M_{v1} - M_{v2} = 96.75 - 2.92$

$$= 93.83 \text{ kg cm}$$

$$M_b = (M_h^2 + M_v^2)^{\frac{1}{2}}$$

$$= \left[(167.58)^2 + (93.83)^2 \right]^{\frac{1}{2}}$$

$$= 192 \text{ kg cm}$$

It is safe to assume that the main shaft has to withstand the full 0.1 H.P. of the manual power.

Considering that the planter may be operated at slower speed ,

$$\text{Torque } M_t = \frac{\text{H.P. x } 4500}{2\pi N}$$

$$= \frac{0.1 \times 4500}{2\pi \times 20} = 3.58 \text{ kg cm}$$

For the shaft to be subjected to combined bending and torsional loads

$$T_e = \left[(K_b M_b)^2 + (K_t M_t)^2 \right]^{\frac{1}{2}}$$

Where T_e = Combined Shear Stress

K_b = Combined shock and fatigue factor applied to M_b

= 1.5 (for minor loads)

K_t = Combined shock and fatigue factor applied to M_t

= 1.0 (for Minor Shock loads)

$$\therefore \quad T_e = \left[(1.5 \times 192)^2 + (3.58)^2 \right]^{\frac{1}{2}}$$

$$= 288 \text{ kg cm}$$

Now, diameter of shaft $d = \sqrt[3]{\frac{T_e \times 16}{f_s \times \pi}}$

$$= \sqrt[3]{\frac{288 \times 16}{600 \times \pi}} = 1.3 \text{ cm}$$

3.7 Design of Intermediate shaft

This shaft is connected to the ground wheel shaft through the chain drive. One 22 teeth sprocket is bolted to one end of this shaft and one bevel pinion of 22 No. of teeth is keyed to the other end. The forces acting on this shaft being identical to the forces acting on ground wheel shaft, the shaft is taken to be 13 mm diameter.

3.8 Design of shaft of Metering Plate

The metering plate is connected to the bevel gear set by a shaft. This shaft is keyed to the bevel pinion at one end and bolted to the metering plate at the other end. This shaft revolves at a speed of

$20 \times \frac{5}{8} = 12.5$ rpm., where the ground wheel rotates at 20 rpm.

So, torque $T = \frac{H.P. \times 4500}{2 \pi N}$

$$= \frac{0.1 \times 4500}{2 \times \pi \times 12.5} = 5.73 \text{ kg cm}$$

Diameter of the shaft

$$d = \sqrt[3]{\frac{T \times 16}{f_s \times \pi}}$$

$$= \sqrt[3]{\frac{5.73 \times 16}{600 \times \pi}}$$

$$= 0.36 \text{ cm}$$

3.9 Design considerations for the Bevel Gears

Bevel gears are used for connecting two shafts which are at right angles. They may be connected at any desired angle. Two types of bevel gears are used - straight tooth gears and spiral tooth gears. In case of straight tooth gears the elements of the teeth converge to a common point called the apex.

The definitions relating to bevel gears are given below.

Pitch line : The pitch line is a straight line passing through the apex of the cone and lying on the slant surface.

Pitch angle : The pitch angle is the angle between the pitch line and the axis of the cone.

Addendum : The addendum is the distance by which tooth extends outside the pitch line at the outer edge while the dendum is the depth of the tooth below the pitch line at the outer edge.

Root angle : The root angle or the cutting angle is the between the tooth base line and the cone axis which intersect at the apex.

Face angle : The face angle is the pitch cone angle plus the addendum angle.

Pitch diameter : The diameter of the base cone is known as the pitch diameter.

The specifications for a gear drive include the horse power to be transmitted the speed of the driver, the velocity ratio to be transmitted and the centre distance.

The following points should be taken into considerations while designing a gear drive for a particular service.

(i) The gear should have sufficient strength so that it does not fail at starting torques under dynamic running conditions.

(ii) The teeth must have very good wear characteristics so that the life of the gear is long.

(iii) The suitable material combination must be chosen so that it gives good wear characteristics and the drive is silent.

(iv) The drive should be compact.

(v) The drive should be properly aligned

(vi) The proper lubrication arrangements should made.

3.10 Design of Bevel Gears

This gear box is intended to give power to the metering device through a pair of bevel gears which would have a ratio of 5 : 8. The gear combination of 10 teeth pinion and 16 teeth bevel and adopting a suitable module m = 5 mm., the detailed design is given below :-

3.10.1 Pitch Circle diameter

Pitch Circle diameter of pinion $A = \left(\frac{n \times m}{10}\right)$

$$\text{Or, } D_1 = \frac{10 \times 5}{10} = 5 \text{ cm}$$

Pitch circle diameter of wheel $8 = \frac{N \times m}{10}$

Or, $D_2 = \frac{16 \times 5}{10} = 8$ cm

The bevel gear rotates at a speed of rpm and the pinion rotates at rpm.

Therefore velocity ratio

$$= \frac{\text{teeth on the bevel}}{\text{teeth on the pinion}} = \frac{16}{10} = 1.6$$

$$\tan\delta_2 = \frac{1}{V.R.} = \frac{1}{1.6} = 0.625$$

$$\text{Or, } \delta_2 = \tan^{-1} 0.625 = 32^\circ$$

$$\therefore \delta_1 = 90^\circ - 32^\circ = 58^\circ$$

3.10.2 Pitch line velocity

Pitch line velocity V_m is given by

$$V_m = \frac{\pi D_2 N}{60 \times 100} \text{ m/sec} = \frac{\pi \times 8 \times 38}{60 \times 100}$$

$$= 0.159 \text{ m/sec}$$

The gears are to be designed for in maximum load of 0.1 h.p. So, the transmitted tooth load of this speed is

$$F_t = \frac{H.P. \times 75}{V_m} = \frac{0.1 \times 75}{0.159} = 47 \text{ kgf}$$

3.10.3 The design stress

The materials for the gears are cast iron grade 35 for which the ultimate tensile strength δ_u = 3500 kg

Taking factor of safety $n = 2.5$

The design tensile stress $= \frac{3500}{2.5} = 1400$ kg

3.10.4 The dynamic load on bevel gears

The dynamic load on bevel gears is given by the equation.

$P_d = C_v \; N_{sf} \; K_m \; F_t$

P_d = dynamic load in kgf

C_v = velocity factor = $\dfrac{3.5 + v_m^{\frac{1}{2}}}{3.5}$

$= \dfrac{3.5 + (0.159)^{\frac{1}{2}}}{3.5}$

$= 1.11$

N_{sf} = service factor = 1.5

K_m = load distribution factor

= 1.25 since both gears over-hang

F_t = 47 kgf

So, P_d = (1.11) (1.5) (1.25) (47) kgf

= 97.8 kgf

3.10.5 Calculation of circular and diameter pitch

The circular pitch $= \pi m$

$= \pi \times 5 = 15.7$ mm. $= P_c$

The diameter pitch $= 1/m = 1/5 = 2$ mm $= P_d$

3.10.6 Determination of Face Width

Adopting a 20^{6} involute teeth for the gear Lewis form factor based on a circular pitch is given by -

$$y_v = 0.154 - \frac{0.912}{Z_v}$$

Also for straight bevel gears

$$Z_v = \frac{Z_1}{\cos\delta_1} \quad \text{or} \quad \frac{Z_2}{\cos\delta_2}$$

$$\text{or, } Z_v = \frac{10}{\cos 58^\circ} = 18.87$$

$$\therefore \quad y_v = 0.154 - \frac{0.912}{18.87} = 0.106$$

$$Y_v = \pi \times y_v = 0.33$$

Applying Lewis' equation ,

$$F_s = \frac{(\delta_b)\ b\ Y_v\ (1 - \frac{b}{R})}{P_d}$$

Where δ_b = design bending stress kgf/cm^2

= 600 kgf/cm^2 for C.I grade 35

b = 0.3 R = face width for tooth

Y_v = 0.33

P_d = 0.2 cm

F_s = strength of tooth in kgf = 97.8 kgf

$$\therefore \quad 97.8 = \frac{600 \times b \times 0.33\ (1 - \frac{0.3R}{R})}{0.2}$$

$$= \frac{600 \times b \times 0.33 \times 0.7}{0.2}$$

$$\text{Or,} \quad b = \frac{97.8 \times 0.2}{600 \times 0.33 \times 0.7} = 0.14 \text{ cm}$$

Since this value of face width is very small the face width may be assumed as 0.3 R or $\frac{10}{P_d}$

$$R = \text{cone distance} = 0.5\ M_t \sqrt{Z_1^2 + Z_2^2}$$

Where $M_t = M_{av} \dfrac{\gamma_y}{\gamma_y - 0.5}$

Taking the value of $\gamma_y = 3$

and $M_{av} = m = 5$

$$M_t = \frac{5}{10} \times \frac{3}{3 - 0.5} = \frac{15}{2.5 \times 10} = 0.6 \text{ cm.}$$

Therefore $R = 0.5 \times 0.6 \sqrt{10^2 + 16^2}$
$= 5.66$ cm

Therefore $b = 1/3\ R = \frac{5.66}{3} = 1.89$
$= 2$ cm (say)

3.10.7 Safe cone distance

Now, checking the safe cone distance R

$$R \geqslant \gamma_y \times \sqrt{i^2 + 1} \quad \sqrt[3]{\left(\frac{0.72}{(\gamma_y - 0.5)K_a}\right)^2 \frac{E\ M_t}{i}}$$

$M_t = M \times K_d \times K$

Where $K_d \times K = 1.3$ for symmetric scheme

and $M = 3.58$ kg cm

$M_t = 1.3 \times 3.58 = 4.654$ kg cm

Therefore

$$R \geqslant = 3\sqrt{1.6^2 + 1} \quad \sqrt[3]{\left(\frac{0.72}{2.5 \times 6000}\right)^2 \times \frac{1.4 \times 10^6 \times 4.65}{16}}$$

Or, $R \geqslant = 3 \times 1.89 \times 0.21$ $\quad R \geqslant 1.196$

So, R can be conveniently taken as 4.5 cm

3.10.8 Contact Compressive Strength

Now checking the gear for contact compressive strength σ_c is given by the formula -

$$\sigma_c = \frac{0.72}{(R - 0.5b)} \sqrt{\frac{\sqrt{(i^2 + 1)^3}}{i \times b} E (M_t)}$$

$$\sigma_c = \frac{0.72}{4.5 - 1} \sqrt{\frac{(1.6^2 + 1)^{3/2}}{1.6 \times 2} \times 1.4 \times 10^6 \times 4.654}$$

$$= 760.77 \text{ kg/cm}^2$$

But the design tress for cast iron grade 35 gear is -

$$\sigma_c = 6{,}000 \text{ kg/cm}^2$$

As the actual value of σ_c is less therefore the design is safe.

3.10.9 Bending Stress

Also checking the bevel pinion for design bending stress. The bending stress for straight bevel gear is given by -

$$\sigma_b = R \sqrt{\frac{(i^2 + 1) \qquad (M_t)}{(R - 0.5b)^2 \; b \; m \; Y_v}}$$

$$= 4.5 \sqrt{\frac{(1.6^2 + 1) \qquad (4.654)}{(4.5 - 0.5 \times 2)^2 \; 2 \times 5 \times 0.33}}$$

$$= 0.975 \text{ kg/cm}^2$$

It is much less than the designed bending stress of the material which is 3200 kg/cm^2. Hence the design is safe.

3.10.10 Check for dynamic load

The pinion can be checked for dynamic load by Buckingham's equation -

$$F_d = F_t + \frac{0.164\ V_m\ (c \times b \times F_t)}{0.164\ V_m + 1.485\sqrt{c \times b + F_t}}$$

V_m = pitch line velocity = 0.159 m/sec

C = Constant whose value for 20^0 full depth cast iron gears is given by

C = 5930 e

Where e is the permissible error in action corresponding to module of 5, e = 0.056

$\therefore$ C = 5930 x 0.056 = 332.08 mm

= 33.2 cm

$$F_d = 47 + \frac{0.164 \times 0.159\ (33.2 \times 2 + 47)}{0.164 \times 0.159 + 1.485\sqrt{33.2 \times 2 + 47}}$$

$$= 47 + \frac{1.836}{12.496}$$

$$= 47.15$$

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It is below the maximum permissible contact compressive stress of 6000 kg/cm^2. So the design is safe.

3.10.11 Check for maximum wear

The limiting load on the basis of wear of pinion is given by formula -

$$F_w \quad d_1 \; Q \; K \; b$$

Where F_w = wear load kgf

d_1 = pitch circle diameter

Q = ratio factor

$$= \frac{2\,i}{i + 1}$$

$$= \frac{2 \times 16}{1.6 + 1}$$

$$= 1.23$$

k = load factor

$$= \frac{(\sigma_c)^2 \; \sin\alpha \, (1/E_1 + 1/E_2)}{1.4}$$

Where α = pressure angle = 20°

σ = design compressive strength for the material of the gear = 6000 kg/cm^2

E_1 = E_2 = 1.4×10^6

$$K = \frac{(6000)^2 \sin 20^\circ}{1.4 \times 1.4 \times 10^6}$$

$$= 6.28 \ kg/cm^2$$

$$\therefore \quad F_w = 5 \times 1.23 \times 6.28 \times 2$$

$$= 77.24 \ kgf$$

Now, F_w is greater than F_d

From Buckingham's dynamic load equation the wear is within the limit.

3.10.12 Dimensions for the bevel gear set

Pinion

Outer Diameter	=	5 cm
Face Width	=	2 cm
Reference angle	=	32°
Number of teeth	=	10

Gear

Outer diameter	=	8 cm
Face Width	=	2 cm
Reference angle	=	58°
Number of teeth	=	16

CHAPTER - IV

MATERIALS AND METHOD

The major considerations in the fabrication of an implement and machinery are the cost of fabrication, the strength, durability, operational safety and easiness in operation, availability of spare parts. These objectives can be fulfilled with proper selection and quality of material used in building it. The fabrication, operation and adjustments are made simple such that the farmer can repair and maintain the implement easily.

4.1 Fabrication of the Field Model

For convenience of fabrication, the groundnut planter has been divided into six major components -

i) Metering Device
ii) Drive Mechanism
iii) Ground Wheel
iv) Furrow opener
v) Seed hopper and seed tube
vi) Frame

The fabrication procedure of each of the component is dealt with in detail in the following sections of the chapter.

4.1.1. Metering Device

The metering plate is made of cast iron having 15 cm. diameter. Four notches have been provided along the periphery of the plate. The size of each notch is 8 mm. x 8 mm.

The base of the hopper holds the metering plate. The base is made of cast iron. The inside diameter is 15 cm. The metering plate rotates inside the base freely. The metering plate is connected to the small bevel pinion by one 9 mm. diameter mild steel shaft. The plate is fixed to the shaft by two nuts.

4.1.2 Drive Mechanism

The drive mechanism consists of two components, the chain drive and the gear drive. The power from the ground wheel is conveyed to the bevel gear set by the chain drive. One pinions of 22 teeth has been fixed at the shaft of the ground wheel by 6 mm set screws. Another Intermediate shaft has been fixed parallel to the ground wheel shaft on the metering box base plate. This shaft runs through one M.S. bush of 5 cm length. The two shafts are made of Mild Steel having 13 mm diameter. At one end of the Intermediate shaft, the 22 teeth sprocket has been fixed by 6 mm. set screws. To the other end of this shaft the bevel pinion with sixteen number of teeth has been fitted. A Rolon No. G-52 chain having 8.0 mm. pitch has been used.

4.1.3 Ground Wheel

The ground wheel has been made of finely dressed sand stone. This has been made with a view to finalise the diameter of the ground wheel after through test in the field

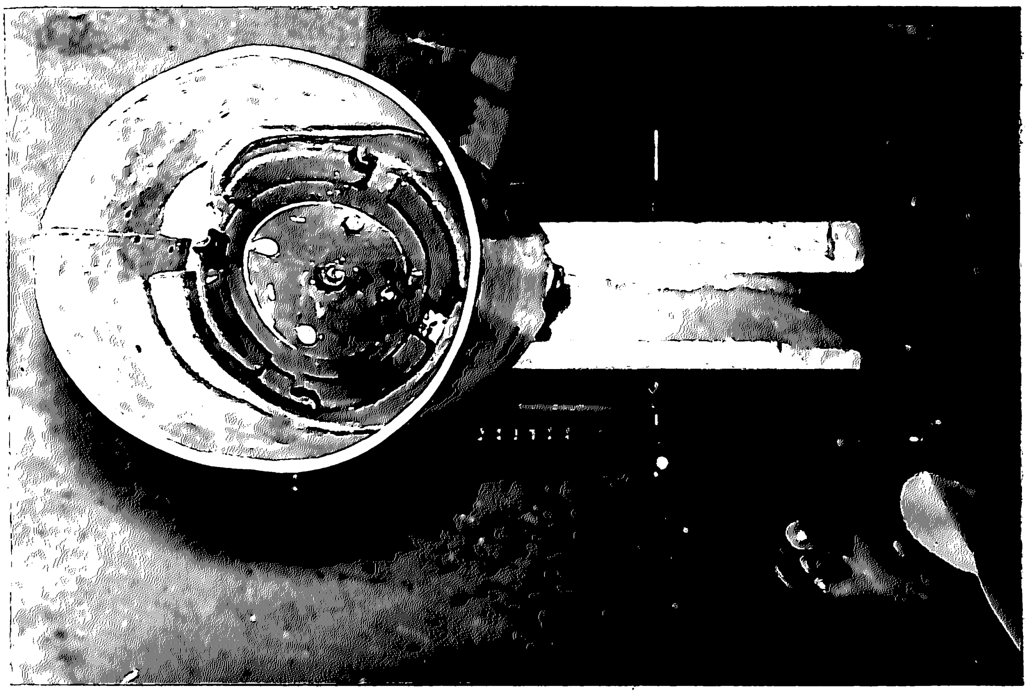

Fig. 4.2 Metering Device of Groundnut Planter.

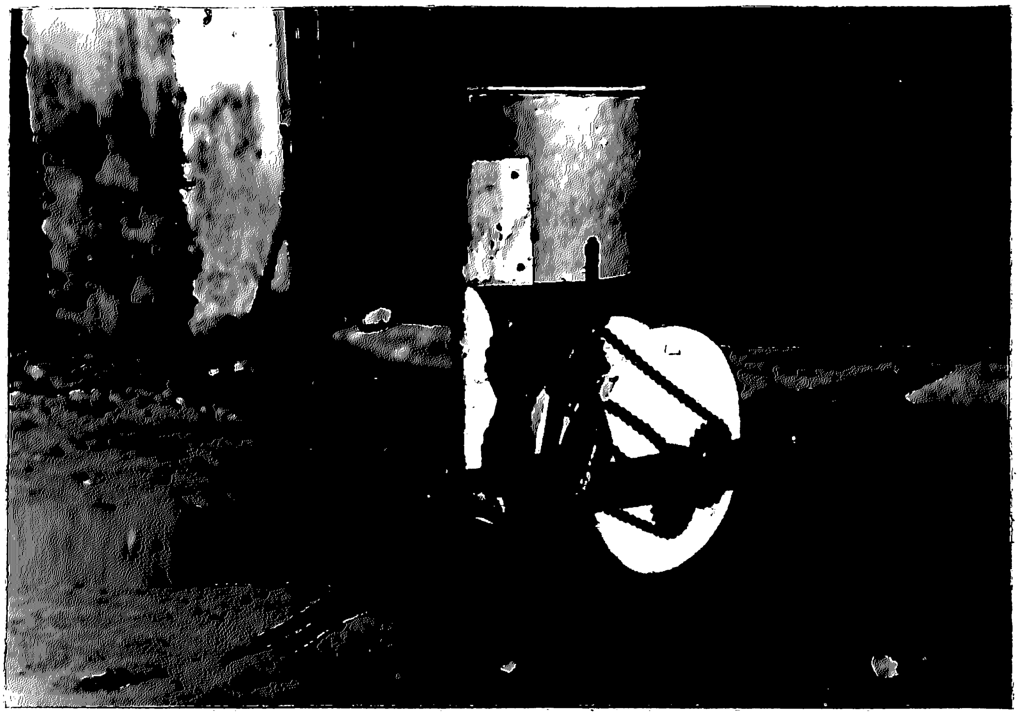

Fig. 4.3 The Groundnut Planter.

considering actual slip occuring. The circumference of the wheel is 96 cm. Both the edges of the wheel have been made to slope inward, the inside diameter being 26 cm. This has been made to make the wheel stable as well as to provide better compaction of soil around the seed. The mild steel shaft has been fixed to the ground wheel by keys.

4.1.4 Furrow Opener

The furrow opener has been prepared by 30 x 30 mm angle. The seed tube made of M.S. Sheet of 1.6 mm. thick and is welded to the furrow opener. The furrow opener is fixed to the frame by nuts and bolts at an angle of 50° to the vertical.

4.1.5 Seed Hopper

The seed hopper is made of G.P. Sheet of 24 SWG thickness. It has a capacity of 3 kgs. of kernel. It's height is 20 cm. and the base is fixed to the metering mechanism base plate by means of dowels and two nos. of 5/16"x3/8" countershunk nut bolts. A parts of the hopper extends up to the furrow opener functioning as the seed tube.

4.1.6 Frame

The purpose of the frame is to hold the different components of the planter together. The frame has been made with 30 mm. x 5 mm. size M.S. Flat. The bushes for the ground wheel has been welded. So also the supporting stand for the seed hopper has been welded to the frame. The total length of the frame is 64 cm. One wooden handle has been attached to the front end to facilitate pulling.

4.2 Test Procedure

The developed planter was calibrated in the laboratory and tested on the soil bin.

4.2.1 Laboratory calibration

The developed planter was mounted on a stand. The ground wheel was rotated for 100 revolutions. The seeds coming out from the seed tube were collected in a polythene bag. The number of seeds were then counted and from this seed to seed spacing was calculated. It was observed that no seed was damaged by the metering device. This test was repeated for 10 times and the data collected in a table.

4.2.2 Laboratory test

In the soil bin, the soil was loosened by hoeing and then levelled. A length of 10 metres was marked on the soil surface. The planter was filled up with seeds and run over the distance.

The planter was run ten times and from each run, a sample length of 1 metre was taken at random. The spacing of hills, number of seed of hill and deviation of seed from the centre line of travel were recorded and analysed.

CHAPTER - V

RESULTS AND DISCUSSION

This chapter deals with the results obtained from the tests conducted and the discussions made there-on.

5.1 Laboratory Calibration of Groundnut Planter

The result obtained from the test is tabulated in the table No. 5.1. The average spacing between hill to hill is 14.54 cms. in place of designed spacing of 15 cms. This shows a variation 3 % of the designed value. This variation occures due to the following reasons.

i) More than one seed is dropped in some of the droppings.

ii) Nearly 1 % of seeds are damaged by the planter.

5.2 Plant to Plant spacing of Groundnut Seeds

As presented in table 5.2, the spacing of seeds in the furrow varies from 14 cm to 16 cm. The average spacing comes to be 15.07 cm. The test of significance for large samples was done taking the sample mean as 15 cm, which is the designed spacing. The Z value comes to be 1.13 which is not significant. Hence, the mean spacing of 15 cm. from hill to hill is satisfactory.

5.3 Placement of Groundnut Seeds in Rows

The placement seeds from the centre line of travel of the planter is to recorded in table 5.3. The placing varies from 0 to 3 cm. away from the centre line. The average displacement is 0.8 cm.

The Z test is made an assumed mean displacement of 0 cm. and the corresponding value of Z becomes 6.02, which shows that Z is significant. Hence taking a mean displacement of 1 cm., the Z value comes out to be 1.50, which is not significant. So it is concluded that definitely there would be some displacement. This displacement is caused due to the clearance between the seed and seed tube.

5.4 Dropping of Groundnut Seeds in Hills

The number of seeds dropped per hill was noted in table 5.4. It is observed that number of seeds per hill varies from 1 to 3. The average being 1.25 Nos.

The Z test is conducted on the sample taking a sample mean of 1.00 seed per hill, which is the designed value. The Z value comes out to be 4.02 which is significant. So the sample mean of 1 cannot be accepted. Then a sample mean of 1.2 was tested and the Z value comes out to be 0.8 which is not significant. Hence the sample mean of 1.2 Nos. fits the test. The deviation from the designed value is attributed due to following causes.

i) Variation in size of seeds.

ii) Jerks associated with the travel.

Table : 5.1 Laboratory Calibration of Groundnut Planter

Sl. No.	No. of revolutions	Movement of Ground wheel	No. of seeds collected	Damaged seeds	Average seed to seed spacing cms.	Remarks
1.	50	4800	340	6	14.37	$\bar{x}$ = 14.54
2.	50	4800	335	5	14.54	n = 10
3.	50	4800	336	6	14.54	σ = 0.18
4.	50	4800	342	6	14.28	taking μ = 15 cm.
5.	50	4800	332	3	14.58	t = 8.08 (significant)
6.	50	4800	335	4	14.50	
7.	50	4800	326		14.91	taking μ = 14.50
8.	50	4800	328	4	14.81	t = 0.7 (not significant)
9.	50	4800	338	5	14.41	
10.	50	4800	336	4	14.46	
Total :	500	48000	3348	33	14.54	

Table : 5.2 Plant to Plant spacing of Groundnuts, Cm.

Rows	Spacing, cm.						Remarks
1.	14	15	15	15	15	16	
2.	15	15	15	16	14	15	
3.	15	16	16	15	15	15	
4.	16	15	14	15	15	15	
5.	16	15	15	15	15	14	$\bar{x}$ = 15.07 σ = 0.48
6.	15	15	15	15	15	15	For μ = 15
7.	16	15	16	15	15	15	Z = 1.33 (Not significant)
8.	15	15	15	15	15	15	
9.	15	16	15	15	15	14	
10.	15	15	15	15	15	15	

Table : 5.3 Placement of Groundnut Seed in rows

Rows	Displacement from centre of travel, cms.							Remarks
1-	2	0	2	3	2	0	0	
2-	0	2	2	0	1	0	1	
3-	3	0	0	2	0	0	0	
4-	0	0	0	0	0	2	1	$\bar{x} = 0.8$
5-	0	0	0	0	0	0	0	
6-	2	3	0	0	1	0	0	$\sigma = 1.11$
7-	4	0	0	1	2	2	0	taking $\mu = 0$
8-	1	2	1	0	0	3	0	$Z = 6.02$
9-	0	0	0	0	0	0	0	(significant)
10-	1	2	3	0	2	3	0	taking $\mu = 1$
								$Z = 1.50$ (non-significant)

Table : 5.4 **Dropping of Groundnut Seeds in hills**

Rows	Dropping, no.							Remarks
1.	1	1	1	2	1	1	2	$\bar{x}$ = 1.25
2.	2	1	1	1	2	1	1	σ = 0.52
3.	1	1	1	1	1	1	1	For, μ = 1.00
4.	1	2	2	3	1	1	1	Z = 4.62 (significant)
5.	1	1	1	1	1	1	1	
6.	2	1	1	2	1	1	2	For, μ = 1.20
7.	1	1	1	1	1	1	1	Z = 0.8 (Non significant)
8.	1	1	2	1	1	1	2	
9.	2	1	3	1	1	1	1	
10.	1	1	1	1	3	1	1	

CHAPTER- VI

SUMMARY AND CONCLUSION

6.1 Summary

Increasing the production of oil seeds in the country can save import of edible oils. Groundnut is one of such oil seeds whose production is steadily increasing. This shows the increased acceptance by farmers to the groundnut as cash crop. This is a crop which can be grown in both khariff and rabi seasons and with irrigation and without it.

Mechanising the cultivation will increase the production and expansion of its area. Manual labour is cheap and plenty in India. Manually operated implements are readily accepted by the small and marginal farmers of the country, their percentage being very high.

Therefore, attempt has been made to design and develop a groundnut planter. This would be cheap, easy to operate and serviceable by village artisans.

The groundnut planter is a single row, manually operated. The power for the metering device is obtained from the ground wheel in two steps. The primary drive transmits power from ground wheel to the intermediate shaft by a Rolon G 52 chain having 8 mm. pitch. The two sprockets have 22 teeth each. The secondary drive consist of the intermediate shaft, a bevel pinion set of 1.6 gear ratio and the metering shaft. The metering plate is fixed to the metering shaft. The metering plate rotates inside the base of the seed hopper. The plate is of 15 cm. diameter. It has four notches of 8 mm. square . The ground wheel is 30 cm. dia and is made of sand stone. The seed hopper is made of G.P. Sheet and is of 3 kg. capacity. The seed tube is also made of G.P. Sheet. The furrow opener is made of mild steel. All this components have been mounted on a frame of 30 x 5 mm. M.S. Flat. The metering plate rotates at an angle of 60° to the horizontal. The planter is designed to drop one seed at 15 cm. spacing.

This groundnut planter was fabricated and tested in the laboratory. The calibration test shows that there are multiple seed drops, which is due to improper grading of seeds. Due to shortage of time the planter could not be tested in the field. The planter was tested on the soil bin. The results show that hill to hill spacing is maintained at 15 cm.. Placement of seed around the line of travel is within the limit of furrow opener and the ground wheel. Multiple drops occure which are due to un-graded seeds and jerks coming from the soil.

6.2 Conclusion

A manually operated groundnut planter has been designed and developed. This can be used by the small and marginal farmers for efficient planting of groundnut seeds. In a 8 hour day 0.32 hectare of land can be planted. The multiple drops of seed will help maintaining plant population despite failure of germination due to bad quality seed and seed damage.

CHAPTER - VII

SUGGESTION FOR FUTURE WORK.

Design and Development of Groundnut Planter has shown that there is enough scope for future work and the following suggestions are made.

(1) Present system only plants groundnut seeds. This can be provided with a system to drill fertilizer also.

(2) The present groundnut planter is designed for one row planting. This can be developed to be a multi-row planter.

(3) The metering plate having three notches may be provided for planting spreading type groundnut at 20 cm. interval.

(4) The metering mechanism can suitably be modified to plant other seeds like maize, arhar, etc.

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